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No. 787

FACTORS AFFECTING HEAT TRANSFER IN

THE INTERNAL-COMBUSTION ENGINE

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SUMMARY

A method was developed for the direct measurement of the average heat-transfer coefficient from the gases in the cylinder during the cycle of operation of an internal-combustion engine. Experimental measurements were made with a heat collector projecting through a spark-plug hole into the combustion chamber of the test engine, in order to examine the effects of several engine-operating and design parameters on the mean heat transfer to the collector.

For an engine of a fixed compression ratio and valve timing, operating with a given mixture ratio, with ignition adjusted to give maximum pressure at the same crank-angle position, the mean heat transfer was found to be a function of the air consumption only, no matter what changes the volumetric efficiency, inlet-air density, or mean piston speed might undergo. The heat transfer was found to vary with the 0.5 power of air consumption in both a slow-speed and a high-speed C.F.R. engine. This figure was surprisingly constant for all tests.

The effect of the heat-resistive coating formed by the combustion deposits on the surface of the heat collector was also examined. Starting with a clean heat collector, the heat transfer fell about 20 percent during the first 15 hours of operation, after which it fell off at a very slow rate, becoming constant after about 40 hours of operation.

INTRODUCTION

There are several methods for evaluating heat losses in the internal-combustion engine. These methods may be broadly divided into two categories: one based upon the

thermodynamic analysis of indicator diagrams (references
1 and 2), and the other based upon the measurement of the
total amount of heat absorbed by water jackets (references
3, 4, and 5) or cooling air (reference 6). The changes
that actually occur in the engine cylinder are so compli-
cated in nature that no method of theoretical analysis or
experimental investigation may claim to have yielded en-
tirely satisfactory results.

Investigators of the first category often contend
that the direct measurement of heat absorbed by jacket
water or cooling air has no actual bearing on the thermo-
dynamics of the engine, as an unknown proportion of ex-
haust heat absorbed by the exhaust port and its surround-
ings, as well as the uncertain amount of piston-friction
heat, inevitably creep into the measurement. This argu-
ment is quite correct. However, the analysis based upon
the indicator diagram does not appear to give results
which finally justify the amount of labor called for, and
it does not give a good picture of the mechanism of heat
transfer and factors controlling same, so that the prac-
tical application of such analysis is rather limited. It
would be of great thermodynamic interest if sufficiently
reliable indicator diagrams were obtainable at high speeds,
so that instantaneous gas temperatures could be computed
from these diagrams without serious error.

The present investigation is an attempt to evaluate
the effects of air consumption, mean piston speed, mix-
ture ratio, and compression ratio on the mean heat trans-
fer from the hot gases to a heat collector screwed into
the combustion chamber of the spark-ignition engine. Heat
exchange between the cylinder walls and the heat collector
was reduced to the possible minimum by the provision of a
dead-air space around the heat collector, and by equaliz-
ing the temperatures of the two bodies. The interesting
features of this arrangement are:

(1) Heat transfer from the exhaust system was absent.

(2) Heat transfer under conditions of sonic velocity
     during the exhaust "blow-down" period was al-
     most completely absent.

(3) Effect of piston-friction heat was almost com-
     pletely excluded.

In other words, the heat transfer determined by this method corresponded to the loss during the thermodynamic cycle of operation. Also, the labor and inaccuracy of the indicator-diagram method was saved.

The author desires to take this opportunity to express his thanks to Professors C. F. Taylor and E. S. Taylor, of the Massachusetts Institute of Technology, for the advice and help they have rendered. He is also indebted to Professor A. R. Rogowski, Mr. W. A. Leary, and Mr. C. H. Wang for their assistance at various times.

GENERAL EQUATION OF HEAT TRANSFER

In the evaluation of the heat transfer from the hot cylinder gases to the cylinder, it is necessary to consider the relative proportions of heat transmitted by radiation and convection. An accurate estimation of radiation is extremely difficult (references 7, 8, 9, 10, 11, and 12); but from what is known it appears certain that the heat transfer due to radiation is very small. In the modern high-speed internal-combustion engine, in which heat transfer takes place mainly by forced convection, radiation may be taken as negligible.

For the heat transfer by means of forced convection, it may be shown (reference 13) that, for a given engine

$$Q = K_1 (T_g - T_w) L^2 (\rho s)^n \left(\frac{L}{\mu}\right)^{n-1} \qquad (1)$$

where

Q mean heat transfer per unit time

T_g effective gas temperature

T_w average cylinder-wall temperature

ρ effective gas density

μ effective gas viscosity (absolute)

L a characteristic length of the engine

s mean piston speed $(2 S \bar{N})$

S   stroke

N   engine speed

K_1, n   constants

Neglecting the variation in μ, which is probably small for ordinary operating conditions, equation (1) may be written

$$Q = K_2 L^{1+n} (\rho\ s)^n (T_g - T_w) \qquad (2)$$

But

$$\rho = K_3 (e\ \rho_i)$$

Hence

$$Q = K_4 (e\ \rho_i\ s)^n L^{1+n} (T_g - T_w)$$

$$= K_4 (e\ \rho_i\ s\ L^2)^n L^{1-n} (T_g - T_w)$$

$$= K_5 G^n L^{1-n} (T_g - T_w) \qquad (3)$$

where

e   volumetric efficiency[*]

ρ_i inlet-air density

G   air consumption per unit time

K_2, K_3, K_4, K_5 constants

It is of interest to note that the heat transfer from the hot gases to the cylinder is a function of air consumption, no matter what changes the volumetric efficiency, inlet-air density, or mean piston speed may undergo.

Equation (3) may be written in a more generalized form to include some of the important design and operating parameters:

[*] Volumetric efficiency is herein defined as the ratio of the weight of air actually taken into the cylinder to the weight of air which would fill the piston displacement at the inlet density.

$$Q = K_5 \, G^n \, L^{1-n} \, (T_g - T_w) \, f(r, \alpha, \tfrac{F}{A}, \theta) \qquad (4)$$

in which

f indicates an unknown function

r compression ratio

α valve timing

$\dfrac{F}{A}$ fuel-air ratio

θ ignition advance

In equation (4) the effect of the quantity θ is to alter the instantaneous values of gas temperature, gas pressure, gas density and viscosity, etc., all of which are functions of the indicator diagram. It will be noted with reference to figure 1 that for different engine-operating conditions the indicator diagrams are very similar in stroke if the ignition advance is set to give pressure peaks at the same crank-angle position. Thus the ignition advance giving the pressure peak at the same crank-angle position would appear to be the rational basis of comparison.

In other words, equation (3) is valid for geometrically similar engines of the same compression ratio and valve timing, operating with the same mixture ratio, with ignition adjusted to give maximum pressure at the same crank-angle position.

APPARATUS AND EXPERIMENTAL NOTES

This investigation covers tests on a slow-speed and a high-speed C.F.R. engine, of maximum permissible speeds of 1800 and 3600 rpm, respectively. In the slow-speed C.F.R. engine a constant mixture ratio could not be very well maintained in a 3-minute run, on account of the unsatisfactory fuel-supply system. In the high-speed C.F.R. engine, a rather elaborate scheme was employed, in which the fuel was injected into a vaporizing tank by means of a pump operating at 25 pounds per square inch supply pres-

sure. The injection nozzle was made to deliver fuel at a
pressure of 1200 pounds per square inch, the spray angle
being 12° and the atomization good. The fuel was then
mixed with air in the vaporizing tank, which was steam-
jacketed to insure good vaporization. With this arrange-
ment it was possible to obtain an almost perfectly homo-
geneous mixture to be taken into the cylinder. Reference
14 contains a more complete description of the inlet sys-
tem. The mixture remained constant for practically any
length of time by keeping the fuel-supply pressure as well
as the mixture temperature constant. Figure 2 shows the
high-speed C.F.R. engine set-up, and figure 3 shows the
view of the heat collector in position.

Heat collector.- Apart from the complication intro-
duced by the exhaust heat and the piston-friction heat,
difficulty was experienced by previous investigators with
direct measurement of heat given up to the jacket water.
Considerable difficulty was found in keeping the engine-
jacket temperature constant. When the rate of flow of wa-
ter was made large enough to give a steady temperature rise,
the temperature rise was too small for a reasonably accu-
rate estimation of heat transfer. If, on the other hand,
the rate of water circulation was made sufficiently small,
steam "pockets" were formed in the jacket which disturbed
the temperature readings.

This difficulty was overcome by the use of a heat col-
lector shown sectionally in figure 4. A quite similar ar-
rangement had been employed 14 years ago for heat measure-
ment during motoring tests (reference 3); but for reasons
obscure to the present author the work was not carried
further to power runs. In its present version the heat
collector consists mainly of two concentric steel tubes,
put together by a suitable supporting element. Joints
were made watertight by press fitting. No brazing or weld-
ing was used.

The heat collector was then screwed into the combus-
tion chamber of the engine under test through a spark-
plug hole. Cold water was made to enter the inner tube,
while hot water ran out from the outer tube. The rate of
heat transfer was determined by measuring the weight of
circulating water as well as the inlet and outlet water
temperatures.

The area of the portion of heat collector exposed to
the hot gases was 3.75 square inches and its volume, 0.44

cubic inch. Owing to the volume taken up by the heat col-
lector the compression ratio of the engine was altered.
A calibration curve for the various compression ratios
covered by the experiment is given in figure 5.

In order to guard against heat exchange between the
heat collector and the cylinder walls, a dead-air space
was provided in the supporting part of the heat collec-
tor. Clearly, a complete absence of metallic contact was
not possible, and the air in the dead-air space was not
really "dead" in the sense that small-scale turbulence al-
ways existed, hence reducing the effective heat insulation
of the air space. As a necessary precaution the tempera-
tures of the water jacket and the heat-collector outlet
were adjusted to the same value throughout the experiment,
except in the first 40 hours of preliminary runs, in or-
der to eliminate heat transfer from the supporting port
of the collector.

It was remarked that when steam pockets were formed,
the temperature readings became very unsteady, because the
formation of steam bubbles caused the heat-transfer char-
acteristics to change considerably. In the present work
water was kept circulating without stagnation points, and
the outlet temperature of the heat collector was kept be-
low 150° F. There was no evidence of steam formation. A
sufficiently large temperature rise was obtainable, yet
the temperature fluctuation was only of the order of ±0.5°
F in a 3-minute run, corresponding roughly to a half-
percent error. Even when the heat-collector outlet tem-
perature was raised to 180° F, no greater temperature
fluctuation was observed.

Temperature control.- The scheme for the control of
temperature is shown in figure 6. The effective head of
water in the supply tank was maintained constant, and the
rate of flow was adjusted by a needle valve. An extremely
fine control of temperature was obtainable. The engine-
jacket water was circulated by means of a centrifugal pump,
cold water being admitted for temperature adjustment.
With this arrangement, the engine-jacket temperature could
be maintained at ±3° F from the heat-collector outlet tem-
perature in a 3-minute run.

In a work of this kind, it is imperative that a per-
fect temperature equilibrium be reached prior to taking
formal readings. From 1 to 1½ hours would generally be
required for an engine starting from cold. About 15 min-

utes were allowed when engine conditions had been altered.
The oil temperature was not specially controlled but was
always around 140° F.

Air measurement.- In this work an NACA Roots-type
supercharger was used as an air meter. Calibration against
a Durley orifice showed that one revolution of the meter
corresponded to 0.180 cubic foot of air. Surging in the
intake system was suppressed by a surge tank of sufficient
capacity.

Fuel-air-ratio measurement.- The mixture ratio was
normally taken from a Cambridge Exhaust-Gas Analyzer. In
tests in which mixture ratio was important, the fuel and
air were timed and the mixture ratio was computed.

On the high-speed C.F.R. engine set-up, an electrical
device was provided for taking the fuel and air readings
simultaneously.

The Cambridge analyzer was found to give readings
sufficiently accurate for most purposes.

Speed measurement.- The engine speed was read by a
hand tachometer in tests on the slow-speed C.F.R. engine.
A Strobotac operating from a controlled 60-cycle line,
was used on the high-speed C.F.R. engine (reference 14).

Procedure.- The major factors, the effect of which
were to be examined, were: volumetric efficiency, inlet-
air density, and mean piston speed. Trials were accord-
ingly made with

(1)   Constant   s   and varying (e ρ_i): by throttling
      the engine while the engine speed was held
      constant.

(2)   Constant ρ_i and varying (e s): by keeping
      the same inlet-air density while the engine
      speed was varied.

(3)   Constant (e ρ_i) and varying s: by throttling
      the engine in such a way that the air consump-
      tion per stroke was the same.

(4)   Varying   s, e, and ρ_i: by varying engine speed
      and engine throttle at random.

The effect of volumetric efficiency alone was not separately examined, owing to the fact that it could not be varied over a sufficiently wide range without having affected the inlet-air density at the same time.

The ignition advance was adjusted to give best power in all runs except those where it was intentionally varied. The best-power-ignition advance was found to give consistently a pressure peak at from 12° to 15° after top dead center. Tests thus made were therefore on a rational basis of comparison.

The effects of mixture ratio, ignition advance, and compression ratio were also examined from a dimensional standpoint.

It is seen from equation (4) that in order to evaluate the effect of one parameter alone, it is necessary to keep all the other parameters constant. For example, in order to examine the effect of the mean piston speed or engine speed, it is not sufficient to keep the inlet-air density constant. Account must also be taken of the variation of volumetric efficiency. In order to examine the effect of mixture ratio, ignition advance, or compression ratio, it is not sufficient to keep the mean piston speed constant; the air consumption must be kept constant so as to make the comparison dimensionally correct. It has appeared to the writer that some authorities have failed to take notice of this point.

RESULTS AND DISCUSSIONS

This report outlines the conditions of heat transfer corresponding to the thermodynamic cycle of operation in a particular region in the engine cylinder. The conditions of heat transfer in another region will necessarily assume a different magnitude but will, it is believed, follow a similar trend.

It will also be seen immediately that there was a remarkable agreement between tests on the slow-speed and the high-speed C.F.R. engine. This fact may be attributed to the similarity in engine design. Whether it will be so in engines of entirely different designs cannot be stated without experimental verification. Nevertheless, as the problem is related to the thermodynamic cycle only, it appears reasonable to expect a materially similar trend.

Heat transfer by radiation.- The heat collector was
completely chromium-plated in order to determine if radia-
tion from the hot gases could be measured. It was at first
thought that if the heat transfer became greater as soot
and lead oxides were deposited on the polished tube sur-
face, the difference in heat transfer could be attributed
to radiation. Subsequently no such indication was observed.
It may be concluded that radiation heat transfer was at
least smaller than the experimental error.

Effect of heat-resistive coating.- In lieu of the
trend as suspected, the mean heat transfer was found to de-
cline very decidedly as a result of these coatings. This
tendency is shown in figure 7, in which the heat flow per
unit time is plotted against hours of operation for the
same amount of air flow. This tendency was found to have
practically ceased after 40 hours of operation. The coat-
ing was 0.002 inch in thickness after about 80 hours of
operation.

Effect of mixture ratio.- The effect of mixture ratio
on the mean heat transfer is shown in figures 8 and 9.
This effect is plainly due to the change in gas tempera-
ture, for in equation (4), r, α, and θ do not enter be-
cause these parameters were kept under dimensionally cor-
rect basis, G was a constant by adjustment, and L for
the heat collector, was also a constant.

The maximum heat transfer was found to occur with a
fuel-air ratio of 0.074, or with a mixture approximately 7
percent richer than chemically correct, which corresponds
to the mixture ratio giving maximum gas temperature.

Effect of ignition advance.- By a similar argument the
effect of ignition advance may also be attributed to the
change of gas temperature. The relation between the mean
heat transfer and ignition advance is shown in figures 10
and 11, for the slow-speed and the high-speed C.F.R. en-
gines, respectively. The experiment covered a range of ig-
nition advance from 0° to 50° crank angle. It was not pos-
sible to advance the ignition to more than 50°, as preigni-
tion set in and the engine ran very irregularly. At 50°
ignition advance the engine missed fire once in about 12
power strokes.

Effect of compression ratio.- Figures 12 and 13 show
that a slight increase in mean heat transfer accompanied
an increase in compression ratio. An increase in compres-

sion ratio is known to raise the compression temperature
and the flame temperature slightly and to lower the expan-
sion temperature and the exhaust temperature to a greater
extent. Hence the total effective heat transfer from the
engine and its exhaust system is ordinarily found to de-
crease with an increase in compression ratio. In this in-
vestigation, however, the heat transfer from the exhaust
system was largely absent. Therefore, it is logical that
the mean heat transfer was found to increase slightly with
an increase in compression ratio.

Effect of throttling.- The effect of throttling, as
shown in figure 14, can be readily explained by equation
(4). Throttling decreases the flame temperature and the
expansion temperature slightly and increases the compres-
sion temperature to a greater extent. The net effect of
throttling on the term $(T_g - T_w)$, is probably small.
The variation of the mean heat transfer with throttling
may be mainly ascribed to the effect of change in density,
which in turn affects the air consumption. The exponent
n was found to be very nearly 0.5.

Effect of mean piston speed.- From equation (3) it
is clear that the effect of mean piston speed on the mean
heat transfer is the same as that of air consumption when
the volumetric efficiency and inlet-air density are kept
constant. The exponent n was also very nearly 0.5. (See
fig. 15.)

Effect of air consumption.- Figures 16 and 17 show
the relation between the mean heat transfer and the air
consumption when the volumetric efficiency, mean piston
speed, and inlet-air density are varied at random, as
well as the compression ratio. Some results from the
first 40 hours of preliminary runs are also included to
show that while the heat-resistive coating affected the
absolute magnitude of heat transfer, it did not alter the
relation between heat transfer and air consumption. The
value of the exponent n was remarkably consistent and
was always of the order of 0.5.

CONCLUSIONS

(1) The method described here enables one to measure
the mean heat transfer corresponding to the thermodynamic
cycle of operation of the internal-combustion engine di-
rectly and with a fair degree of accuracy.

(2) From the hot gases to the cylinder walls, heat transfer may take place by radiation and forced convection. The heat transfer due to radiation could not be detected in this experiment.

(3) The mean heat transfer in a given cylinder due to forced convection, may be expressed by the equation

$$Q = K \ G^n \ L^{1-n} \ (T_g - T_w) \ f(r, \ \alpha, \ \frac{F}{A}, \ 6)$$

The mean heat transfer is a function of the air consumption, no matter what changes the volumetric efficiency, inlet-air density, or mean piston speed may undergo. The exponent n was found to be of the order of 0.5 for the two engines tested, and was very consistent.

(4) The mean heat transfer was found to be a maximum with a mixture about 7 percent richer than chemically correct. It increased with the ignition advance at a gradually increasing rate until the engine ceased to run regularly. It also increased slightly as the compression ratio was raised.

Massachusetts Institute of Technology,
     Cambridge, Mass., September 1940.

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APPENDIX

HEAT TRANSFER AND ENGINE OUTPUT

It can be shown that the indicated horsepower, I, of an internal-combustion engine may be expressed

$$I = K_6 \, G \, f_1 \left(r, \, \alpha, \, \frac{F}{A}, \, \theta \right) \tag{5}$$

For geometrically similar engines of the same compression ratio and valve timing, operating with the same mixture ratio, with ignition adjusted to give maximum pressure at the same crank-angle position, the indicated horsepower is proportional to the air consumption.

Figure 18 shows the relation between the indicated horsepower and the air consumption for the slow-speed C.F.R. engine at three different compression ratios with optimum mixture ratio and ignition advance. There was no valve overlap and the valve timing was the same for the three cases.

Comparing equations (4) and (5),

$$\frac{Q}{I} = K_7 \, G^{n-1} \, L^{1-n} \, (T_g - T_w) \, f_2 \left(r, \, \alpha, \, \frac{F}{A}, \, \theta \right)$$

$$= K_7 \, (e \, \rho_i \, s)^{n-1} \, L^{n-1} \, (T_g - T_w) \, f_2 \left(r, \, \alpha, \, \frac{F}{A}, \, \theta \right) \tag{6}$$

Thus for a given engine the mean heat transfer expressed as a fraction of the indicated output decreases as the output increases. For geometrically similar engines, it decreases as the engine size increases. For geometrically similar engines running at the same piston speed with the same inlet-air density, (Q/I) is proportional to (L^{n-1}) and $(T_g - T_w)$, provided the function f_2 remains unchanged.

Curve	F/A	Load factor	s, ft/min
a	Optimum	1	600
b	Optimum	3/4	900
c	0.091	-	900
d	0.080	-	900

250 lb/in^2

a
b
c
d

0

Atmospheric line
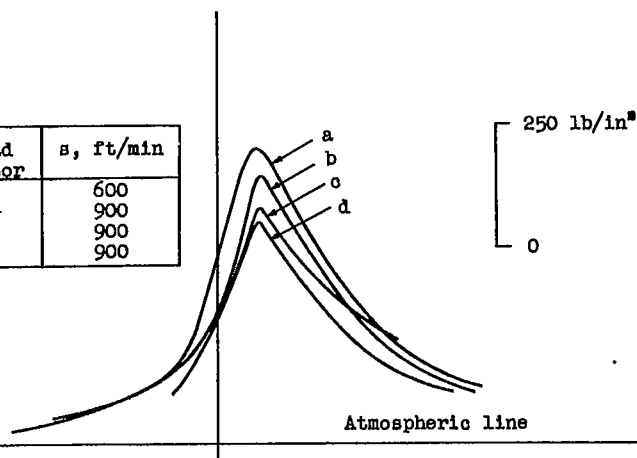
Figure 1.- Indicator diagrams with the best-power ignition advance.
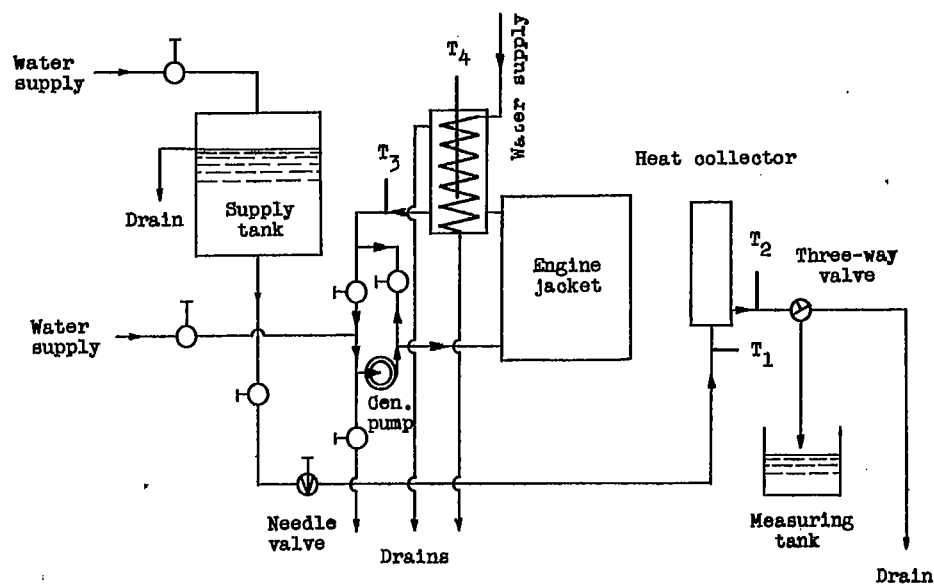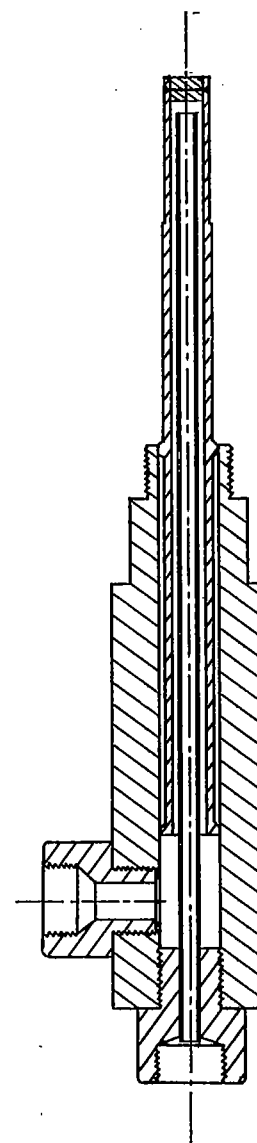100 octane fuel; slow-speed C. F. R. engine.

Water supply

Drain

T_4

T_3

Water supply

Supply tank

Engine jacket

Heat collector

Water supply

T_2

Three-way valve

T_1

Gen. pump

Needle valve

Drains

Measuring tank

Drain

Figure 6. - Schematic piping diagram.

Figure 4. - The heat collector.

Full Size

Vaporizing tank

Fuel and air recorder
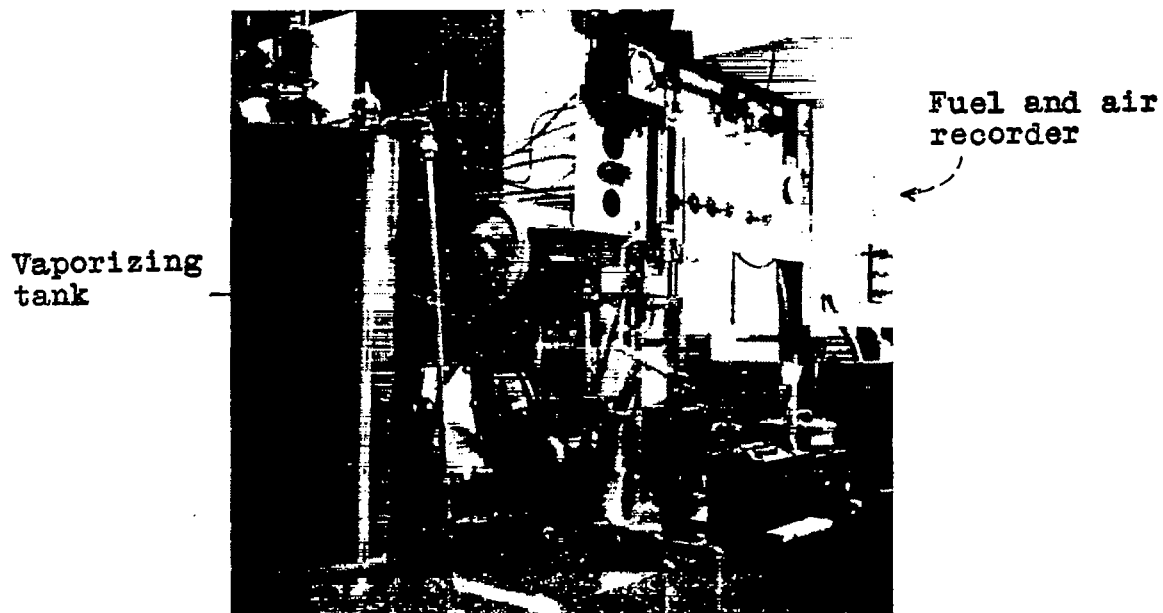
Figure 2. - The high-speed C. F. R. engine set-up, showing the vaporizing tank and the electrical fuel- and air- recording device.
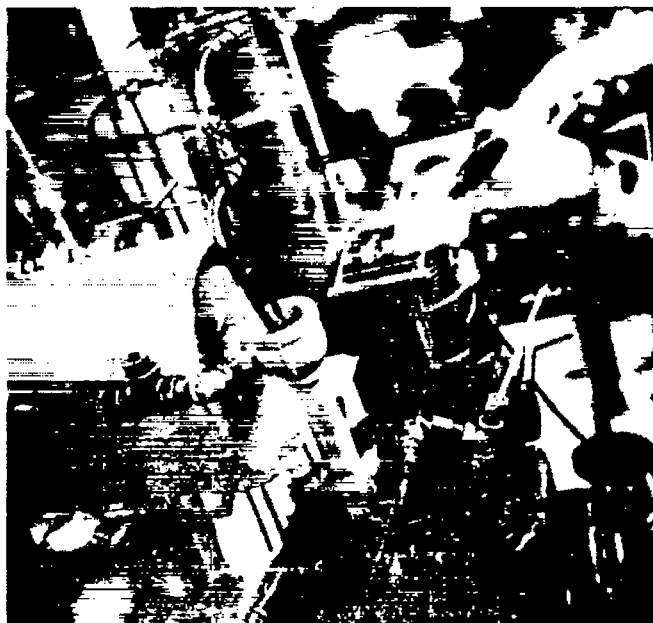
Figure 3. - View of the heat collector in position.
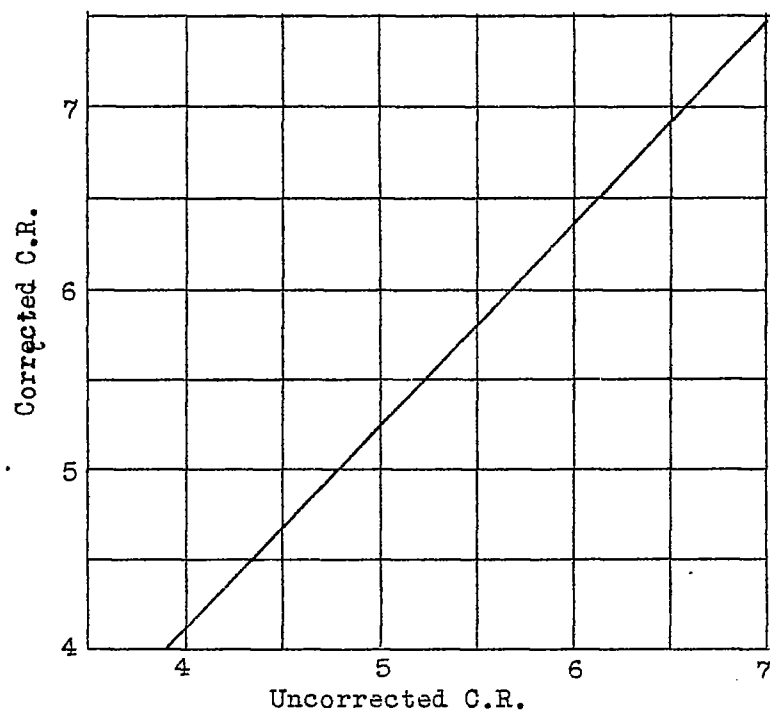
Figure 5.- Corrected compression ratio after insertion
of the heat collector.
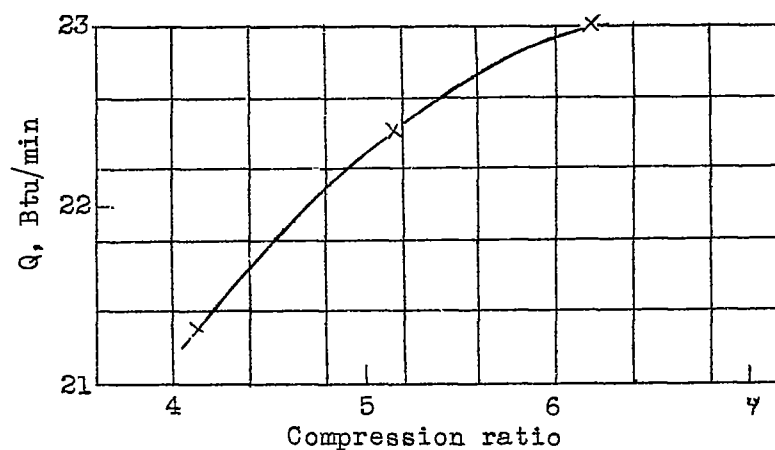
Figure 12.- Variation of heat transfer with compression ratio.
100 octane fuel; optimum θ; $F/A = 0.0757$; $G = 0.80$
lb/min; $T_2 = 145°F.$; $T_3 = 145°F.$; slow-speed C.F.R. engine.
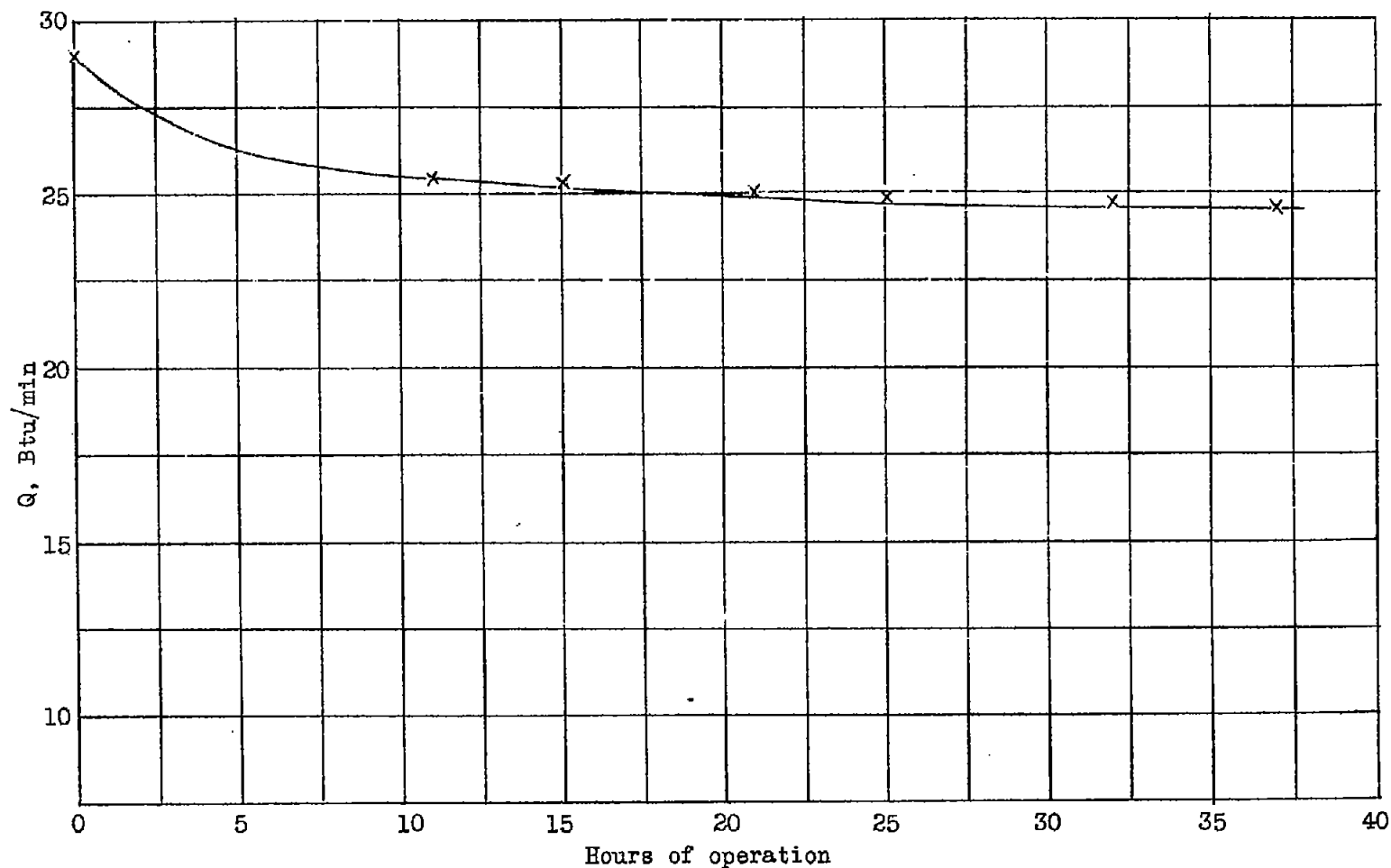
Figure 7.- Effect of heat-resistive coating on heat transfer. Compression ratio = 5.15; optimum F/A and θ; 87 octane fuel; G = 0.80 lb/min; T_2 = 90°F.; T_3 = 210°F.; slow-speed C.F.R. engine.
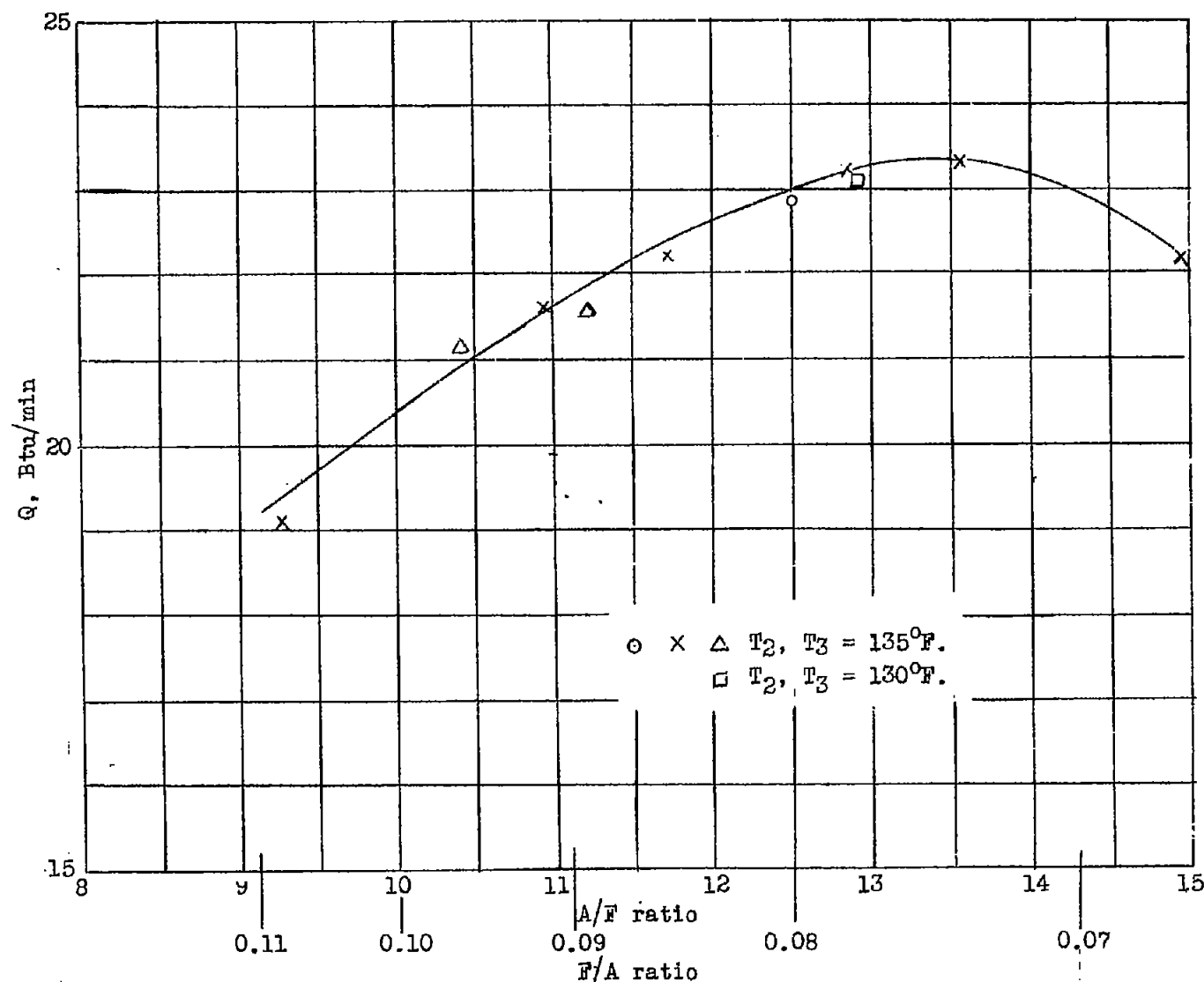
Fig. 7

Fig. 8
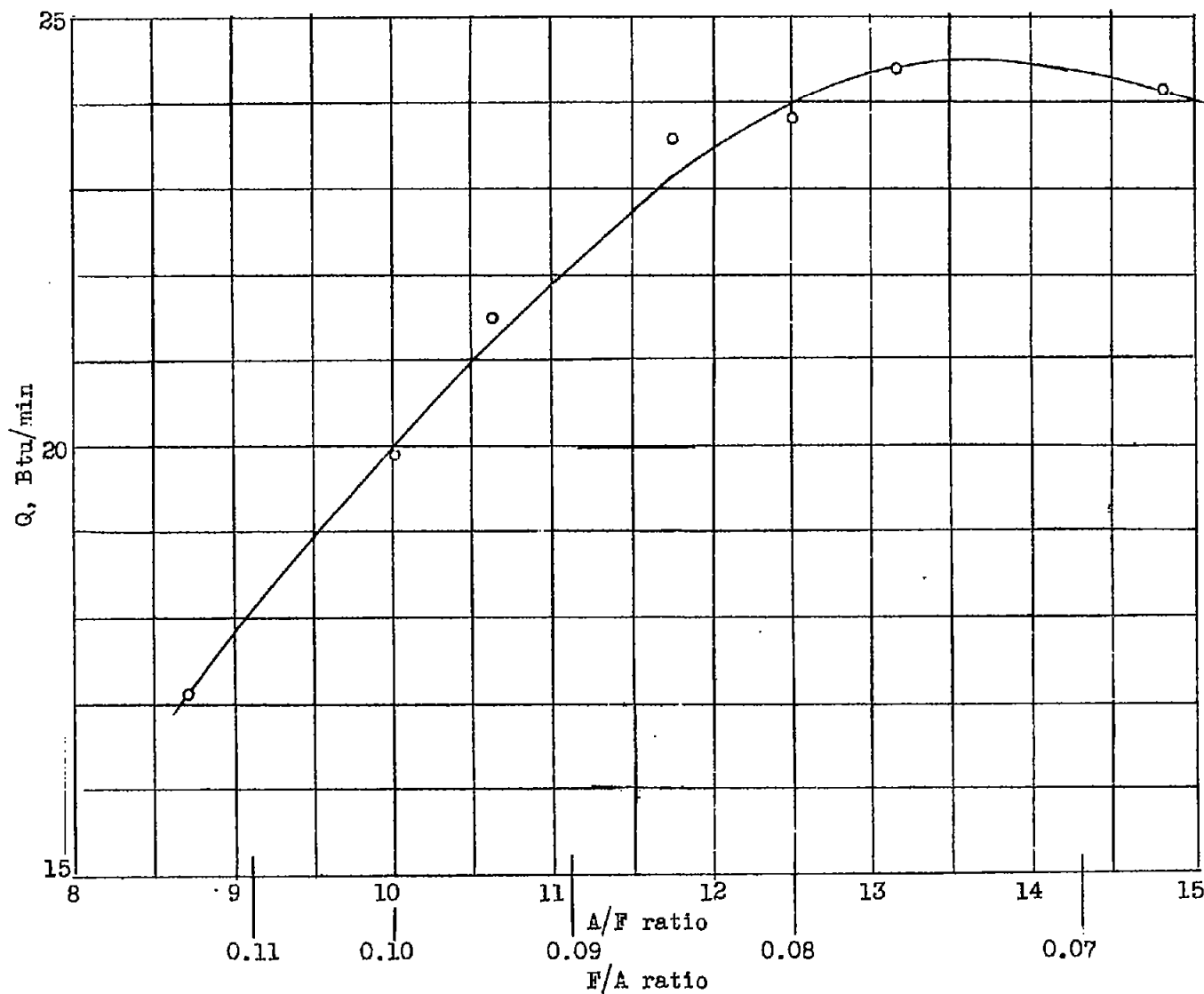
Figure 8.- Variation of heat transfer with mixture ratio. Slow-speed C.F.R. engine; compression ratio = 5.15; optimum θ; 87 octane fuel; s = 915 ft/min; G=0.83 lb/min; T_2=150°F.; T_3=150°F.
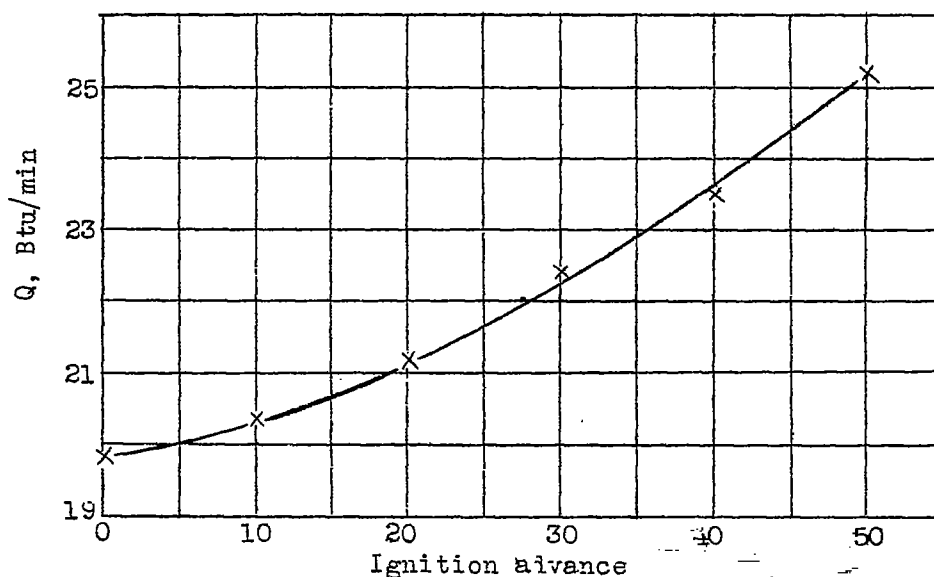
Fig. 9

25

Q, Btu/min

20

.15

8   9   10   11   12   13   14   15

A/F ratio

0.11   0.10   0.09   0.08   0.07

F/A ratio

○ ✕ △ T_2, T_3 = 135°F.

□ T_2, T_3 = 130°F.

Figure 9.- Variation of heat transfer with mixture ratio. High-speed C.F.R. engine; compression ratio = 5.10; optimum θ; 100 octane fuel; s = 900 ft/min; G = 0.76 lb/min.

Figure 10.- Variation of heat transfer with ignition advance.
Slow-speed C.F.R. engine; compression ratio = 5.15;
100 octane fuel; F/A = 0.074; s = 915 ft/min; G = 0.83 lb/min;
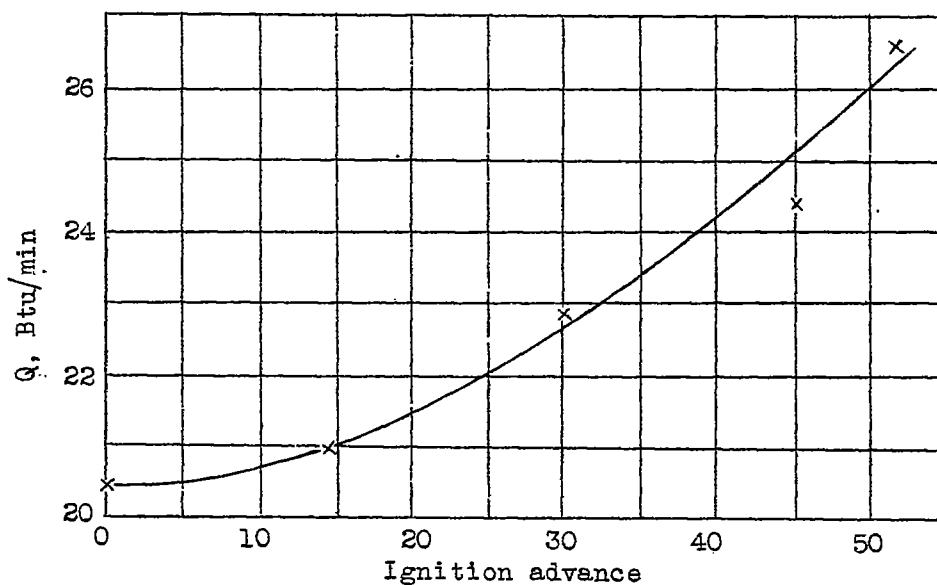$T_2 = 150°F.; T_3 = 150°F.$

Figure 11.- Variation of heat transfer with ignition advance.
High-speed C.F.R. engine; compression ratio = 5.10;
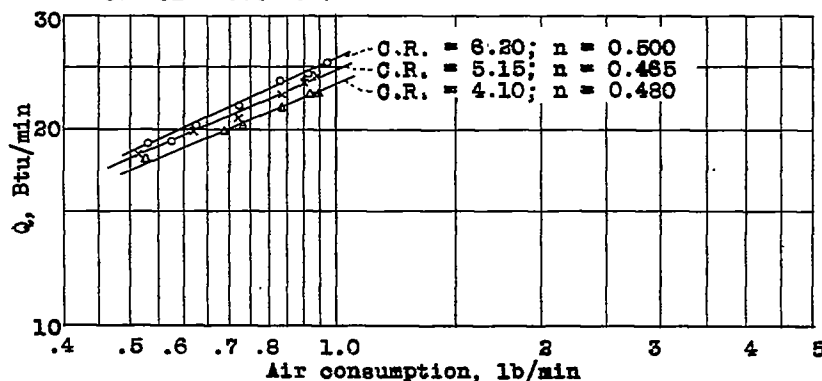100 octane fuel; F/A = 0.080; s = 900 ft/min; G = 0.76 lb/min;
$T_2 = 130°F.; T_3 = 130°F.$

Figure 13.- Variation of heat transfer with compression ratio. 100 octane fuel; optimum θ; F/A = 0.0757; T_2 = 145°F.; T_3 = 145°F.; slow-speed C.F.R. engine.

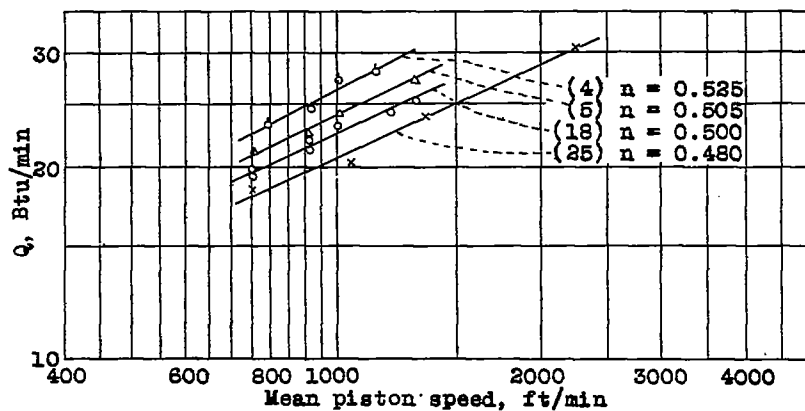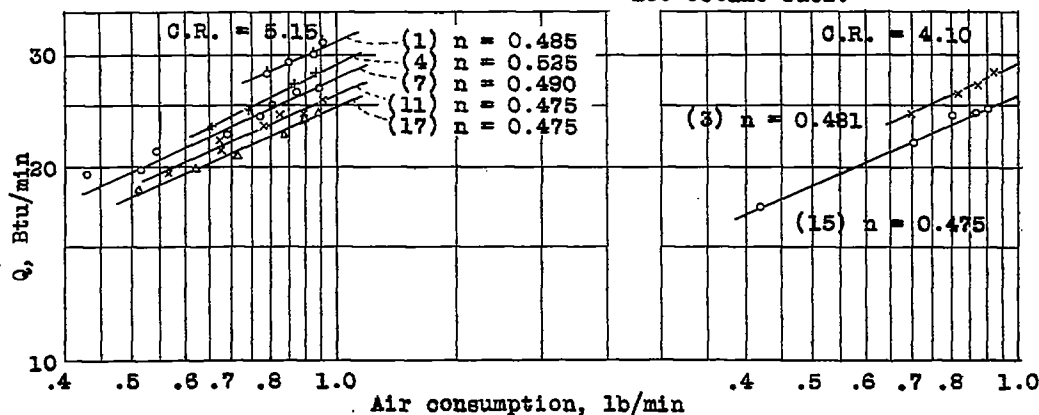| Curve No.	C.F.R. engine	C.R.	T_2, °F.	T_3, °F.	Fuel	Hours of operations
4	Slow	5.15	210	90	87	11
5	Slow	5.15	210	90	87	15
18	Slow	5.15	150	150	100	Over 40
25	High	5.10	140	140	100	Over 40

Figure 15.- Variation of heat transfer with mean piston speed. Optimum F/A and θ; varying mean piston speed.

Curve No.	C.R.	F/A	T_2, °F.	T_3, °F.	Hours of operation
1	5.15	0.080	212	90	0
3	4.10	0.080	212	90	7
4	5.15	0.080	212	90	11
7	5.15	0.080	212	90	21
11	5.15	0.076	212	150	35
15	4.10	0.076	150	150	Over 40
17	5.15	0.076	150	150	Over 40

Figure 16.- Variation of heat transfer with air consumption. Slow-speed C.F.R. engine; varying (e s), (e p1), T_2 and T_3; optimum θ; 87 or 100 octane fuel.

O.R. = 5.10; s = 900 ft/min;
100 octane fuel; F/A = 0.077
optimum O; T_2, T_3 = 130°F.;
High-speed O.F.R. engine.

n = 0.495

O.R. = 6.20; s = 915 ft/min
100 octane fuel; F/A = 0.0757;
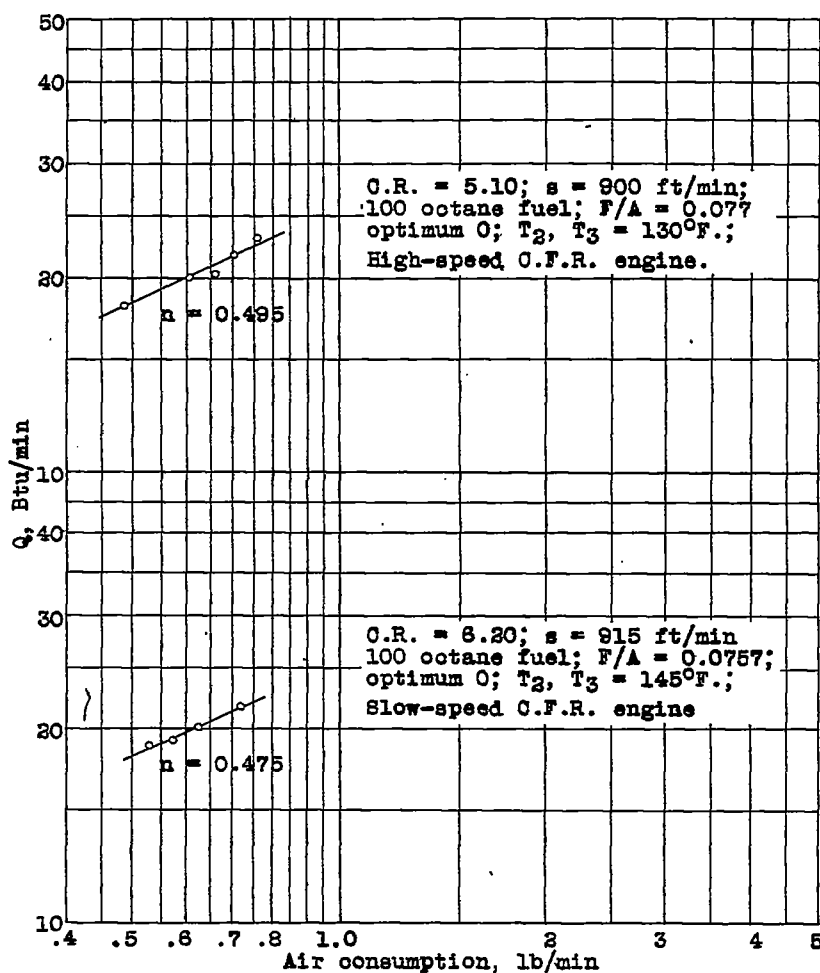optimum O; T_2, T_3 = 145°F.;
Slow-speed O.F.R. engine

n = 0.475

Figure 14.- Variation of
heat transfer
with air consumption due
to effect of throttling
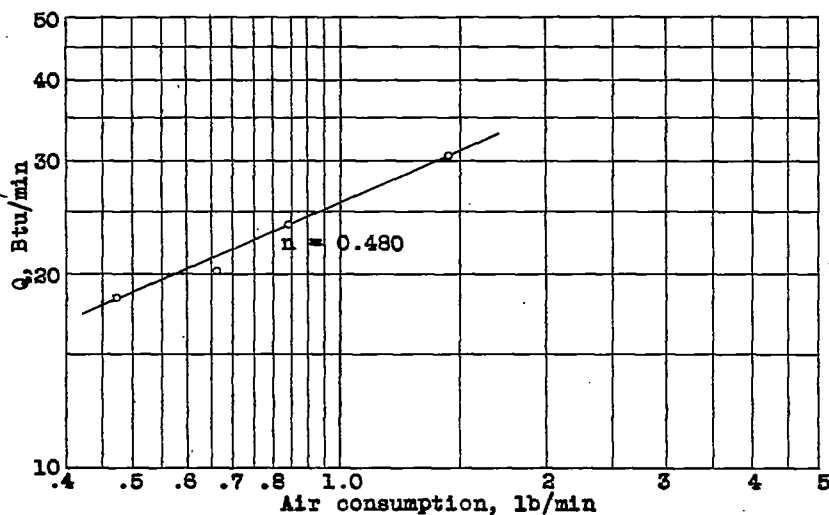at constant piston speed.

n = 0.480

Figure 17.- Variation of
heat transfer
with air consumption.
High-speed O.F.R. engine;
optimum F/A and θ; O.R.=
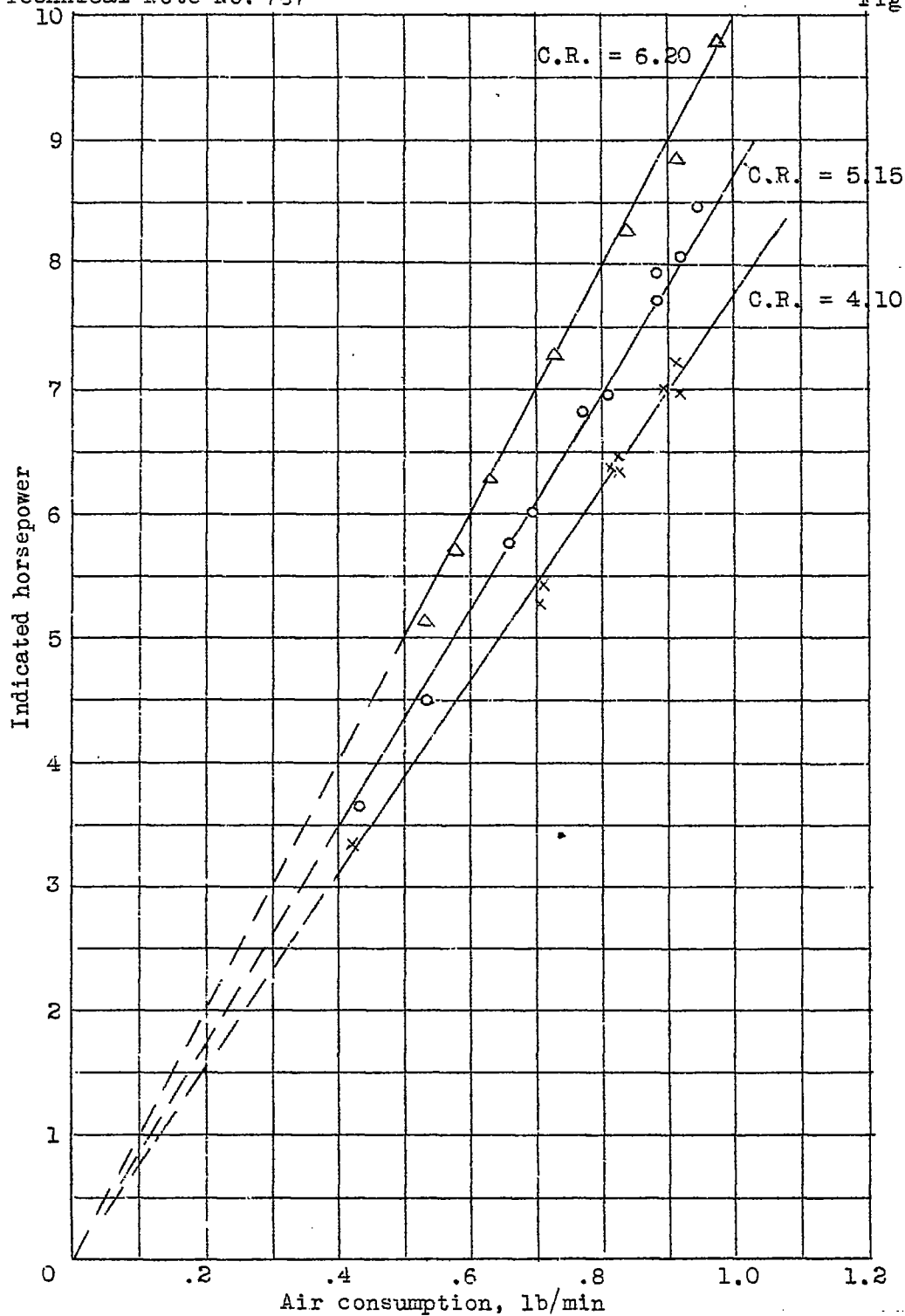5.10; T_2 = 140°F.; T_3 =
140°F.

Figure 18.- Variation of indicated horsepower with air comsumption.
Optimum F/A and θ;varying s, e, and p_i;slow-speed C.F.R. engine.